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# Fatigue life of pitch- and stall-regulated composite tidal turbine blades

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*Summary:* Tidal turbine blades are subject to harsh loading and environmental conditions, including large thrust and torsional loadings, relative to wind turbine blades, due to the high density of seawater, among other factors. The complex combination of these loadings, as well as water ingress and associated composite laminate saturation, have significant implications for blade design, affecting overall device design, stability, scalability, energy production and cost-effectiveness. This study investigates the effect of seawater ingress on composite material properties, and the associated design and life expectancy of tidal turbine blades in operating conditions. The fatigue properties of dry and water-saturated glass fibre reinforced laminates are experimentally evaluated and incorporated into tidal blade design. The fatigue lives of pitch- and stall-regulated tidal turbine blades are found to be altered by seawater immersion. Water-saturation is shown to reduce blade life about 3 years for stall-regulated blades and by about 1 to 2 years for pitch-regulated blades. The effect of water ingress can be compensated by increased laminate thickness. The tidal turbine blade design methodology presented here can be used for evaluation of blade life expectancy and tidal device energy production.

*Keywords:* Tidal energy, tidal blade design, stall-regulated HATT, and pitch-regulated HATT, composite materials, fatigue life.

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## 1. Introduction

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34 Tidal energy is is gaining increased importance as a renewable energy source due to its high  
35 predictability over long timescales [1]. However, the tidal force can vary over a small geographical  
36 space [2], posing a challenge to the reliable long-term design of cost effective structures. The tidal force  
37 can vary locally within distances of tens of meters to tens of kilometres, due to the local bathymetry  
38 and seabed conditions. Hence, blade design requires particular attention. Current tidal turbine (TT)  
39 blade designs are largely based on wind turbine (WT) blade technology. But increasing appreciation of  
40 the specific and unique challenges of the tidal blade environment is leading to specific design evolution  
41 for tidal blades [3]. Tidal blades have to withstand the significant forces of seawater and turbulence  
42 flows, and must withstand water ingress and saturation during the device employment period [4, 5].  
43 The most highly-developed TT technology is the horizontal axis tidal turbine (HATT), which converts  
44 the kinetic energy within the tidal stream into mechanical energy, via the hydrodynamic forces acting  
45 perpendicular to the rotor plane creating blade lift and rotation [5]. In order to stay within generator  
46 capacity, i.e. limit peak power, blade pitching can be used [6, 7]. However, blade failures on a number  
47 of prototypes emphasise the need for a design that will withstand the significant hydrodynamic loads  
48 during expected turbine life.

49 Composite materials, especially glass fibre reinforced polymers (GFRP), are the most commonly  
50 used materials for TT blade design due to their favourable characteristics, e.g. high specific strength  
51 and stiffness, resistance to corrosion and reasonable cost [5, 8]. The importance of environmental effects  
52 on the properties of composite materials has previously been recognised and studied [9, 10]. The  
53 specific application of composites in structural design of ocean energy structures has triggered studies  
54 on immersed performance of composites in seawater [11-13]. Studies show that immersed GFRP  
55 becomes moisture-saturated relatively early in its life. Hence, for TT blade design, it is important to  
56 understand durability and performance of these materials over the device lifetime.

57 The polymers normally used in GFRP can absorb up to 5% water by weight when immersed for  
58 long periods, changing the mechanical properties (e.g. reducing static tensile strength of the material  
59 by more than 25%) [14]. Water diffuses into the polymer matrix [13] and the multidirectional nature of  
60 laminates complicates the fatigue damage mechanisms. Matrix cracking parallel to the fibres or inter  
61 fibre failure (IFF) is first seen in the most off-axis plies under tensile fatigue loading. Most of the IFF  
62 cracking takes place in the first 25% of fatigue life and the significant drop in laminate stiffness is  
63 complete at this stage, with only a minor reduction in stiffness after this point. However, fatigue strength  
64 reductions do not follow the changes in static strength since the damage mechanisms are different in  
65 fatigue [15]. There is little test data available on material behaviour under coupled environmental and  
66 cycling loading [4]. An extensive review of fatigue modelling in GFRP has divided the work among  
67 three broad approaches [16]. First is a testing approach, where life predictions are based on test data of  
68 the exact or a similar material; second is a phenomenological approach, where predictions are based on  
69 the stiffness and residual strength behaviour; third, a progressive damage approach where damage in

[Type here]

70 the unidirectional (UD) lamina is predicted and incremented until a final failure state is reached, thereby  
71 predicting fatigue life. The testing approach to fatigue life estimation is the most widely used [17]. The  
72 technique is under continual evolution and refinement to include effects like spectral loading and  
73 complex constant life diagram (CLD) results [18]. Strength degrades continuously during fatigue and  
74 an early characterization model proposed that it degrades linearly per cycle, in constant amplitude  
75 fatigue [19, 20]. Key problems with all residual strength methods are the large scatter in the residual  
76 strength test results and the complexity of the degradation. The stiffness of GFRP laminates degrades  
77 by between 10 to 20% during fatigue cycling. The technique has been used to predict the life of  
78 particular WT blade laminates.

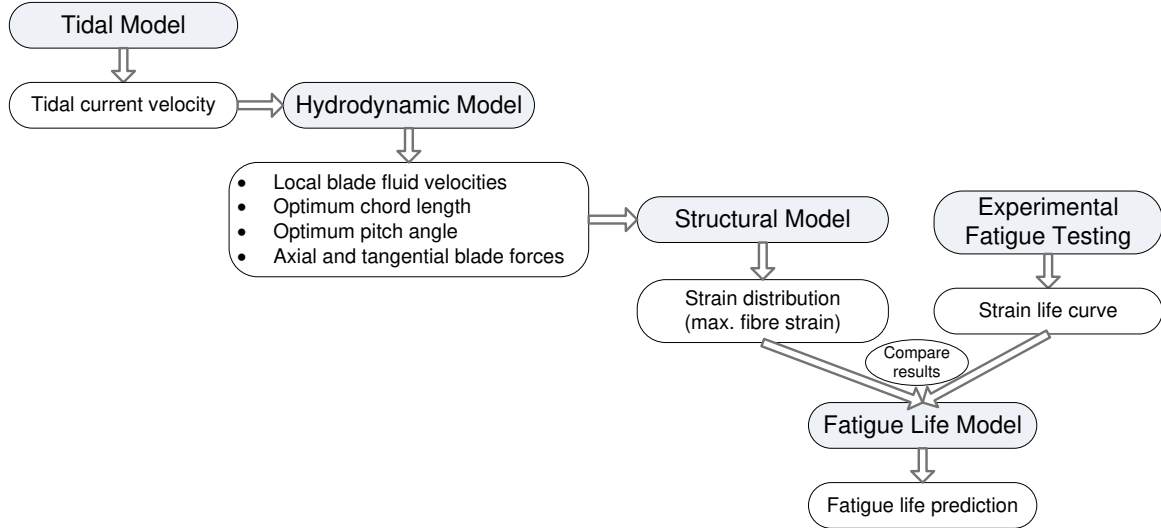
79 The main drawback of the latter models is a lack of flexibility in dealing with different laminate  
80 layups and/or loading patterns. Micromechanical approaches that predict the response of the laminate  
81 based on damage mechanisms in the individual UD plies offer a potential solution. The simplest  
82 approach is to degrade the matrix properties based on observed levels of cracking [21] and use classical  
83 laminate theory (CLT) to integrate the results. Others have considered two damage mechanisms, namely  
84 matrix cracking and interlaminar delamination [22]. Fracture mechanics approaches have been  
85 presented to predict matrix cracking behaviour and fibre failure; energy approaches have been used to  
86 model delamination, with stochastic methods used to enhance existing techniques. However, significant  
87 ongoing work is focussed on improving the capability for predicting test results and to reduce the  
88 amount of testing required to produce reliable fatigue life estimates. In order to fully utilise the TT blade  
89 structural material fatigue life, it is necessary to have information on its performance in the marine  
90 environment for the full design life of marine renewable energy devices (up to 20 years or so). The  
91 literature on fatigue test programs on the use of composite materials in marine renewable energy devices  
92 is, however, limited. Consequently, TT device design tends to be conservative, leading to cost penalties.  
93 A comprehensive fatigue life model for composite blades, incorporating realistic hydrodynamic  
94 loadings, cyclically-varying blade stresses and wet composite material fatigue properties would,  
95 therefore, be valuable tool for TT design.

96 The proposed TT blade fatigue design methodology consists of five modules [23] (Figure 1). The  
97 first module is a tidal model, which predicts the tidal current speed for relevant local tidal velocities  
98 measured [5, 23]. The output from this tidal model forms an input to a hydrodynamic model, which  
99 defines an aerofoil geometry (optimum chord length and pitch angle) and blade loadings (axial and  
100 tangential blade forces). The third module is a structural model. Based on the hydrodynamic module  
101 output, a finite element model of the blade is developed and factored forces are applied in order to  
102 determine the strain distribution in the turbine blade. The fatigue model determines the maximum strain  
103 in the blade for each rotation cycle. The maximum strains are compared to an experimentally-  
104 determined strain-life curve for the material and a damage fraction for that cycle is obtained. Summation  
105 of the damage fractions using Miner's rule gives an estimate for the life of the TT blade (TTB) [24]. In

[Type here]

106 this paper, we adopt this design approach in the context of the water ingression effect on the  
107 performance of stall- and pitch-regulated tidal turbine, in order to integrate fatigue life expectancy of  
108 water-immersed composites into blade design.

109



110

111 Figure 1. Tidal turbine blade design process algorithm.

112

## 113 2. Design Methodology

### 114 2.1 Tidal model

115 It is assumed here that the tidal phenomenon occurs twice within each period of 24 hours, 50  
116 minutes and 28 seconds, consisting of two high and two low tides [25]. The highest tides, spring tides,  
117 occur when the sun and the moon line up with the earth. The lowest tides, neap tides, occur when the  
118 sun and moon are at 90° to each other. The current speed depends on the local topography. However,  
119 if the spring and neap maximum velocities are measured, the full cycle can be approximated by  
120 combining a semi-diurnal sinusoid and a fortnightly sinusoidal function [26]. The tidal current velocity,  
121  $V_t$ , is:

122

$$123 V_t = \cos(\omega_d t)[v_{ave} + v_{alt} \cos(\omega_m t)] \quad (1)$$

124

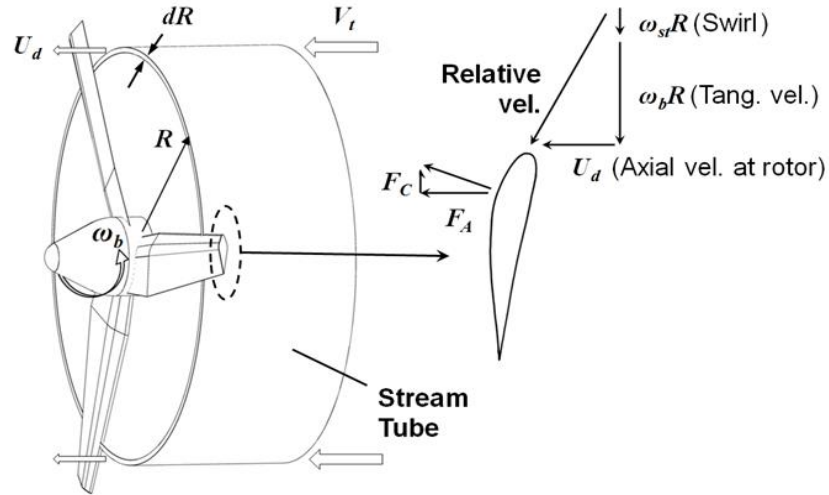
125 where  $v_{ave}$  is the average of the peak tidal velocities,  $v_{alt}$  is half the range of peak tidal velocities,  $\omega_d$   
126 is angular frequency of the tides,  $\omega_m$  is angular frequency of the spring-neap (14.7 day) cycle, and  $t$  is  
127 time.

128

### 129 2.2 Hydrodynamic model (HDM)

[Type here]

130 The design of the aerofoil is dependent on the turbine type and met-ocean condition. The blade  
131 element momentum theory (BEMT) code adopted here for blade design and to predict performance of  
132 HATT blades is based on previous work [27-29]. A stream-tube model (Figure 2) based on BEMT is  
133 employed to calculate steady loads on the turbine blades and the thrust and power of the rotor for  
134 varying fluid velocities, rotational speeds and pitch angles. Optimised chord and pitch angle  
135 distributions can be defined along the span of the blade for a given set of input parameters (Table 1)  
136 [5].  
137



138  
139 Figure 2. Stream-tube model.

140  
141 The stream tube model examines a series of concentric tubes, dividing the blade into a number  
142 of sections, within which momentum is conserved, as it is transferred from the water to the blade. The  
143 BEMT-related mathematical formulation of the forces acting on the blade is given in Appendix 1. The  
144 input data for the HDM in this study is given in Table 1.

145  
146 Table 1. Hydrodynamic model parameters.

Parameter	Value
Water Velocity	2.5 m/s
Water density	1025 kg/m <sup>3</sup>
Water viscosity	0.0013155 Ns/m <sup>2</sup>
Number of blades	3
Revolutions per minute	16
R <sub>outer</sub>	5.0 m
R <sub>inner</sub>	1.5 m
C <sub>L</sub>	1.0
L/D	70
Angle of attack	7°
Z	0.333

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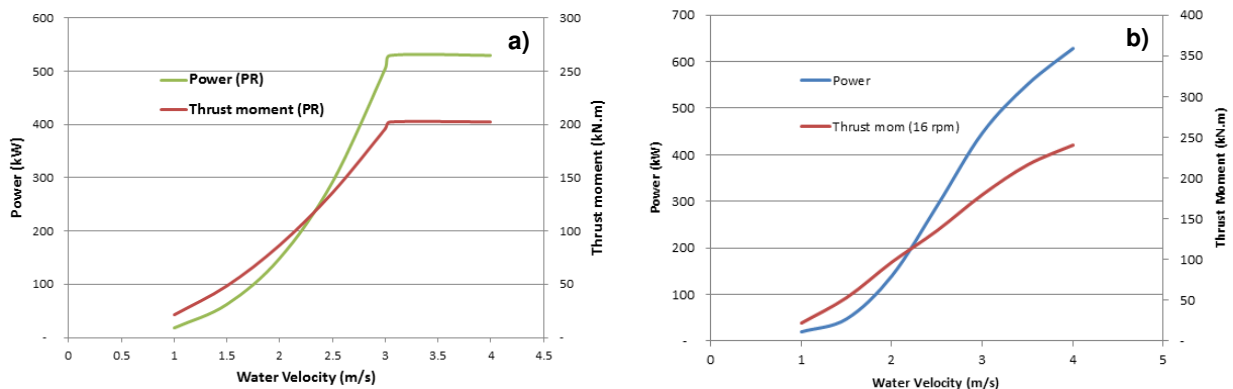
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148 The design of the aerofoil, viz. chord and twist distribution along the blade, is intended to achieve  
149 optimum performance over turbine lifetime. The design code of this paper performs adjustments to the  
150 chord length until moment balance is achieved, after which the process is repeated for all remaining  
151 stream tubes. The outputs at each radial increment are chord length, aerofoil pitch angle, the axial and  
152 tangential force on the blade at particular radial increment, torque, and power.

153 In order to regulate the turbine power during high water velocity, control systems are used to  
154 manage forces and moments on the tidal blade. The HDM is used to simulate the two options for  
155 controlling power, pitch- (PR) and stall-regulation (SR). PR is a system which modifies the lift  
156 coefficient ( $C_L$ ), i.e. the forces on the blade, by rotating the entire blade about its axis. SR blades are  
157 designed with a radial variation of pitch angle so that, as the tidal velocity increases, the angle of attack  
158 over a section of the blade exceeds the stall angle and lift drops off, reducing the forces on the blade  
159 [30]. TT blades have a smaller operational range of velocity, typically 0.75 to 3m/s (exceptionally  
160 4m/s), than WT blades [2, 31]. Therefore, the simpler control systems may be viable for tidal turbines.  
161 For this study the operational environment is such that both control systems produce the same flapwise  
162 moment [32].

163 The maximum theoretical power levels for SR and PR turbines are calculated and the energy  
164 produced by each turbine per year is predicted using HDM (Appendix 1). In order to compare the blade  
165 damage accumulation on two turbines with different control systems, the energy output is matched.  
166 Hence, the aim was to find the tidal current speed at which the energy output of the PR turbine is similar  
167 to that of the SR turbine, to enable an objective comparison with respect to damage accumulation. This  
168 is achieved by identifying the threshold value of water velocity for pitch control, above which blade  
169 pitch is controlled to give constant power. Figures 3a and 3b show the power curves for a PR tidal  
170 turbine (green) and an SR constant speed turbine (blue). Hence, at 3.05m/s current speed these turbines  
171 are determined to have similar energy output in a year, with a power rating is approximately 500kW.  
172

173



173

174 Figure 3. The power curve and thrust moment curve for: a) a pitch-regulated (PR) and b) a stall-  
175 regulated (SR) tidal turbine.

176

177 **2.3 Structural model**

178 The structural design of TTBs is governed by the hydrodynamic shape of the aerofoil and the  
179 extreme loading conditions. The design of an optimised composite TTB requires several iterations  
180 between material characterisation, structural analysis and fluid-structure interaction [4].

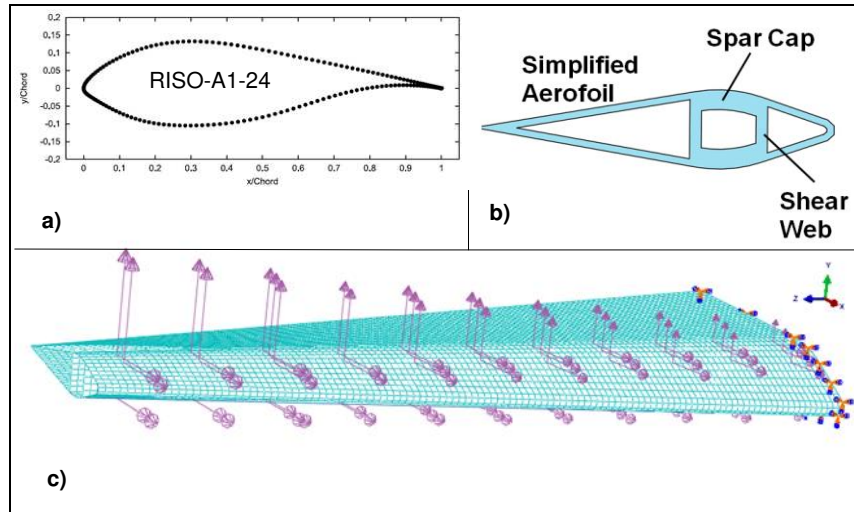
181 The blade cross-sections are selected from the family of shapes that provides the best lift-drag  
182 characteristics [33], which are fairly thin. However, the blade needs to support a complex combination  
183 of loading, including lift, drag, buoyancy, and gravitational forces. These structural requirements lead  
184 to the thicker aerofoil than the hydrodynamic optimum requires [34]. For this study RISO-A1-24 (chord  
185 thickness is 24% of chord length) is identified as the appropriate aerofoil shape for tidal application  
186 [35-37]. The data available for the Risø-A1-24 aerofoil covers a broad range of incidence angles [38].  
187 Hence, the choice of aerofoil is based on hydrodynamic performance and structural characteristics  
188 (Figure 4a).

189 The study examines the fatigue life of a 5.0 m rotor radius, 3-bladed downstream, free-yaw  
190 turbine, which produces approximately 330 kW (operating in a 2.5 m/s tidal current velocity). The  
191 HDM uses 45 stream tubes, uniformly distributed along the blade, and assumes that the water in each  
192 of these tubes has slowed to 1/3 its initial velocity at the exit ( $\zeta = 0.333$ ). The radial distribution of blade  
193 chord length, pitch angle, tangential (torque,  $F_C$ ), and axial (thrust,  $F_A$ ) forces is determined by the HDM  
194 (Appendix 1) for the set of parameters in Table 1. The chord length at the blade root is 1.25 m with a  
195 pitch angle of  $20^\circ$ , while the chord length at the tip is 0.75 m with a pitch angle of  $4^\circ$ . The aerofoil  
196 shape has been simplified in the structural model (Figure 4b).  $F_A$  increases linearly with increasing  
197 radius, from 0.7 kN to 2.3 kN at the extreme blade radius, and is significantly larger than the (power-  
198 generating)  $F_C$ , which increases with radius from 320 N to 410 N at the extreme blade radius. These  
199 forces cause a flapwise bending moment of 150 kNm and edgewise bending moment of 35.7 kNm,  
200 respectively, at a 1.5 m radius from the rotor centre. A finite element (FE) model of a TTB has been  
201 created using Abaqus FEA software [39] (Figure 4c).

202



[Type here]



203

204 Figure 4. Tidal blade modelling: a) Riso A1-24 cross-section [35], b) simplified aerofoil shape used for  
 205 final element (FE) analysis, and c) 5 m long tidal turbine blade FE model with applied loading.

206

207 The parts of the structure bearing the loads and moments (the spar caps) are separated as far apart  
 208 as aerofoil geometry enables, but joined together by two shear webs, creating the box section (the main  
 209 structural element of the blade). The top caps of this box are 35 mm thick at the root and taper to 6 mm  
 210 at the tip. The shear webs and fairings are 12 mm thick and taper to 4 mm at the tip. In the FE model,  
 211 the panels of the blade are modelled as shell elements with quasi-isotropic (QI) material properties  
 212 obtained from the standard laminate analysis [40, 41]. The material characteristics for this study are  
 213 summarised in Table 2. High-stress locations near the surface are modelled with individual plies for  
 214 which properties are calculated from the Rule of Mixtures and the Halpin–Tsai equations [42].

215

216

Table 2. Tidal blade glass fibre/epoxy material properties.

Inputs		Single unidirectional ply properties		Quasi-isotropic laminate [(45\135\90\0) <sub>2</sub> ] <sub>s</sub>	
$V_f$	50%	$E_1$	38 GPa	$E_x, E_y$	19.3 GPa
$E_f$	72.4 GPa	$E_2$	11.6 GPa	$G_{xy}$	7.2 GPa
$\nu_f$	0.22	$G_{12}$	3.5 GPa	$\nu_{xy}$	0.330
$E_m$	3.5 GPa	$\nu_{12}$	0.285		
$\nu_m$	0.35				

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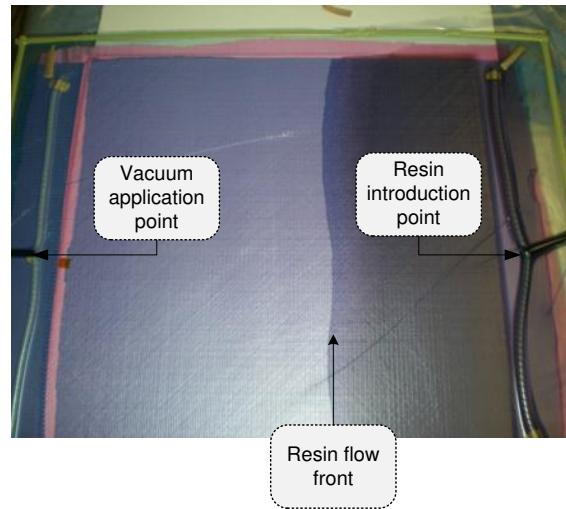
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### 219 3. Experimental method

#### 220 3.1 Coupon manufacture

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221 QI laminates were manufactured using the Vacuum Assisted Resin Transfer Moulding  
222 (VARTM) process [43]. Biaxial stitched glass-fibre mat ( $0^\circ$ -300 g/m<sup>2</sup>,  $90^\circ$ -300 g/m<sup>2</sup>) was cut and  
223 stacked to create a [(45/135/90/0)<sub>2</sub>]<sub>S</sub> laminate and epoxy resins were infiltrated under vacuum (Figure  
224 5). The laminates were cured for 48 hours at room temperature and then post-cured for 5 hours at 80°  
225 C in an oven. The average thickness of the laminate is 3.75 mm with 50% fibre volume fraction. The  
226 laminates were cut into 25 mm × 250 mm coupons for testing.  
227



228  
229 Figure 5. Laminate manufacturing using vacuum assisted resin transfer moulding (VARTM) [43].

230

### 231 3.2 Accelerated ageing procedure

232 The total number of test coupons for material fatigue testing was ten. Five of these were  
233 acceleration-aged in warm water (30° to 40°) for up to 2.5 years (900 days) to simulate immersion in  
234 12° C seawater for approximately 20 years. The rest were stored at normal room temperature and  
235 humidity for a similar length of time. Thus, by increasing the water temperature, movement of moisture  
236 by diffusion into the polymer resin within the composite is accelerated. The diffusion rate ( $k_d$ ) varies  
237 with temperature (Arrhenius law):

238

$$239 \quad k_d = k_0 e^{-\frac{E}{RT}} \quad (2)$$

240

241 where  $k_0$  is the reference diffusion rate coefficient,  $E$  is the activation energy,  $R$  is the universal gas  
242 constant (8.3145 kJ kmol<sup>-1</sup>K<sup>-1</sup>), and  $T$  is the temperature (K). An acceleration factor, for a higher  
243 temperature in the same material, has been defined as [44]:

244

[Type here]

$$F_{H,L} = e^{-\left[\frac{E}{R}\left(\frac{1}{T_H} - \frac{1}{T_{ref}}\right)\right]} \quad (3)$$

246

247

248

249

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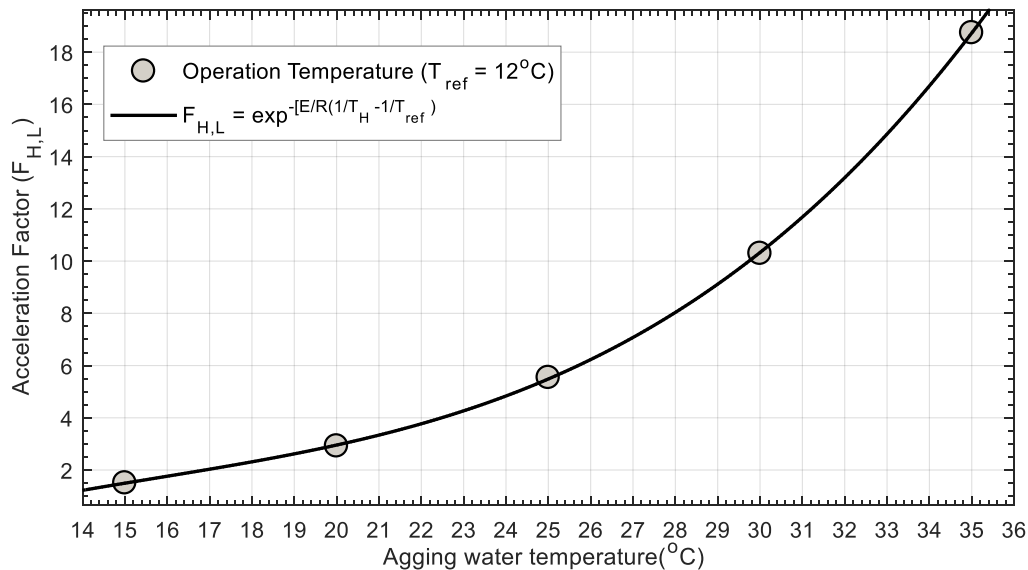
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254

Selection of the higher temperature ( $T_H$ ) to produce a required acceleration factor relative to a given reference ( $T_{Href}$ ) temperature depends on the value of the activation energy ( $E$ ), taken here as 93 kJ kmol<sup>-1</sup> for epoxy/E-glass composite [44, 45]. Figure 6 shows a plot of the acceleration factors for epoxy versus aging water temperature. For a tidal turbine operating off the coast of Ireland, the design operating temperature ( $T_{ref}$ ) is 12°. The figure shows that at 30°C ageing temperature, the acceleration factor is approximately 10 and at room temperature (20°C) it is approximately 3, when compared to the 12°C operating environment.



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256

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Figure 6. Anticipated acceleration factor for epoxy/E-glass composite due to ageing using elevated water temperature.

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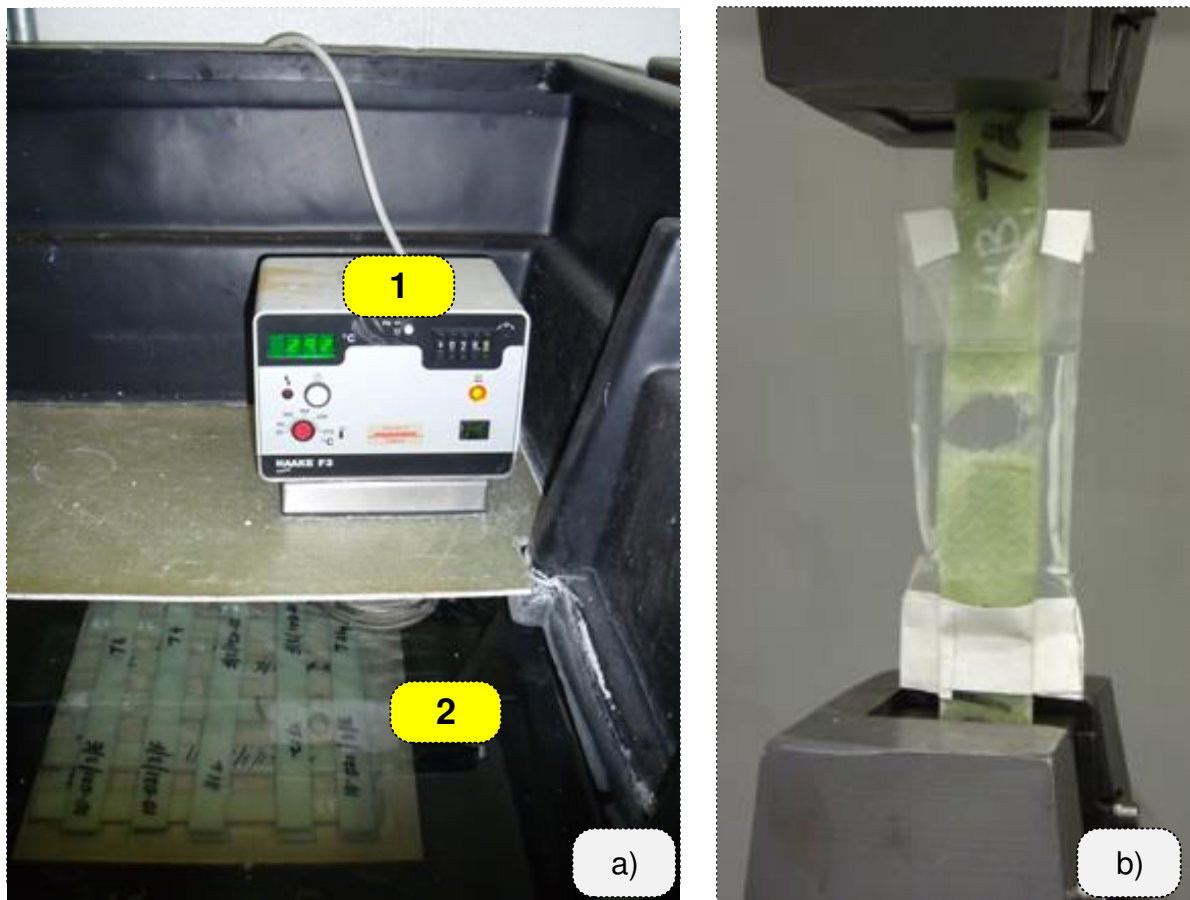
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266

Two containers were filled with tap water and equipped with heater-stirrer units to ensure that the heated water is circulated throughout the entire tank (Figure 7a). The water bath temperature was controlled by a heater and thermometer. The epoxy/E-glass coupons were immersed in baths and maintained at 20°C (unheated tank) and 30°C ± 1°C (heated tank). The coupons are stacked on an aluminium plate in the bottom of the bath in a way that allows maximum water circulation around the coupons while supporting the coupons evenly, to prevent stressing of the coupons during the aging process. Before placement in the tank, the coupons were weighed and the weight recorded to a resolution of 0.001 g. The average weight for the epoxy/E-glass coupons was 43.872 g with a variation

[Type here]

267 of up to 4.1%. To measure the amount of water absorbed, the coupons were re-weighed at intervals  
268 during the immersion ageing process.  
269



270  
271 Figure 7. a) Setup of the ageing water baths: 1. Heater-stirrers used to maintain constant uniform  
272 temperature in immersed ageing water baths and 2. Immersed Epoxy / E-glass coupons; b) Immersed  
273 fatigue test setup, polyethene pouch with waterproof tape.

274

### 275 3.3 Fatigue test method

276 Fibre orientation and volume fraction ( $V_f$ ) are primary determinants of the static and fatigue  
277 strength of a composite [40]. The progressive fatigue zone starts at approximately 80% of ultimate  
278 tensile strength (UTS), which corresponds to around 1000 cycles and reduces to approximately 25% at  
279 over 1 million cycles [46]. Fatigue tests were performed with an Instron 8500 test machine and 8800  
280 controller, applying a sinusoidal force in tension-tension mode ( $R = 0.1$ ) between 3 and 6 Hz. To ensure  
281 that the coupon does not overheat, the fatigue test frequency is decreased to enable a constant rate of  
282 strain application. After the immersion-aged period, the coupons are subjected to sample initial tests to  
283 develop the test setup. Studies show that epoxy composites recover most of their original strength when  
284 they are re-dried after water saturation [11, 47]. In order to keep the samples immersed during the

[Type here]

285 fatigue test, a plastic bag pouch sealed with waterproof tape was used to protect the specimens from  
286 drying (Figure 7b). The majority of the fatigue testing was carried out at a constant amplitude in tension-  
287 tension ( $R = 0.1$ ) mode. However, some coupons were exposed to fatigue cycling between 15.5 MPa  
288 and 155 MPa for  $10^4$  cycles, after which they were tested to failure in a constant amplitude ( $R = 0.1$ )  
289 fatigue test to 180 MPa maximum stress. A power-law relationship (Coffin-Manson equation) is  
290 adopted, relating maximum applied strain ( $\epsilon_{max,i}$ ) and fatigue life:

291

$$292 \quad \epsilon_{max,i} = AN_f^{-B} \quad (4)$$

293

294 where  $N_f$  is the number of cycles to failure, and  $A$  and  $B$  are constants.

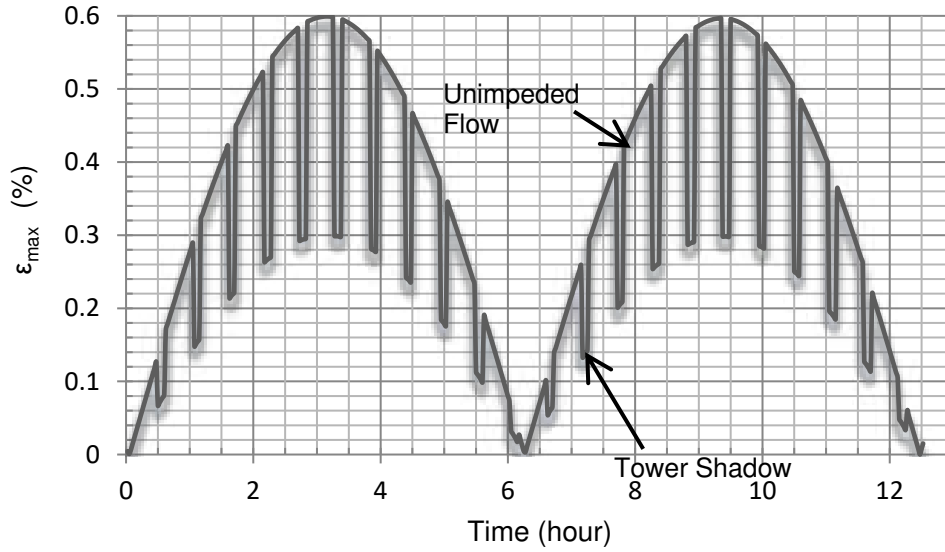
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#### 296 **4. Fatigue Life Model**

297 The focus of the fatigue model is the highest tensile strain in the blade, caused predominantly by  
298 bending. Each tidal cycle causes an increase and then a decrease in the maximum strain on the blade.  
299 Concurrently, for each revolution of the device, the blade experiences a cyclic load caused by the  
300 blocking ('shadow') effect of the support tower. The tower reduces the water velocity downstream  
301 leading to a brief reduction in the lift force and bending moment on the blades in every cycle. [8, 23]  
302 (Figure 8). Wave interaction can also lead to fluctuations in the load on TTBs [3]. The testing of TTs in  
303 wave tanks has found flapwise bending moment amplitude reductions of 50% [48, 49] due to tower  
304 shadow. In this study, a reduction of 50% of TTB bending moment is considered which corresponds to  
305 a fatigue loading R-ratio ( $\epsilon_{min}/\epsilon_{max}$ ) of 0.5, as shown in Figure 8.

306

[Type here]



307  
308 Figure 8. The effect of tower shadow on the maximum strain in the turbine blade (simplified  
309 representation).

310

311 There is no generally accepted method for dealing with the effect of mean stress in fatigue of  
312 composites, i.e. for inferring the  $\epsilon-N$  curves from one (test)  $R$ -ratio to a different  $R$ -ratio [50].  
313 Therefore, to deduce the  $R=-1.0$  and  $R=0.5$   $\epsilon-N$  data from the (measured)  $R=0.1$  data, the method  
314 described in [51] is used, based on the constant life diagrams (CLD) assumed behaviour:

- 315 1) The  $N_f=5000$  line joins the  $R=0.1$  test data point to the material ultimate strain,  $\epsilon_u$ , data  
316 point on the mean strain axis.  
317 2) For  $-1 \leq R \leq 0.5$ , the constant life lines (CLL) are parallel to the  $N_f=5000$  line, based on  
318 experimental observations.  
319 3) For  $R > 0.5$ , the CLLs join the  $R=0.5$  data points to the  $\epsilon_u$  data point on the mean strain  
320 axis.

321 The material of the turbine blade is linearly elastic, hence the major strains in the blade are  
322 directly proportional to the flapwise bending moment, accordingly the maximum strain at any time  $j$  is:

323

324 
$$\epsilon_{max,j} = \epsilon_{max,sp} \left( \frac{M_f(v_j)}{M_{ref}} \right)^2 \quad (5)$$

325

326 where  $M_f(v_j)$  represents the functional dependence of bending moment on tidal current velocity,  $M_{ref}$   
327 is the reference bending moment and  $\epsilon_{ref}$  is the maximum strain in the blade when the reference  
328 moment is applied corresponding to a specific reference velocity  $V_{ref}$ .

[Type here]

329 The fatigue model has two components: I) sums the damage due to the strain cycles caused by  
330 the low-frequency semi-diurnal tidal cycle, and II) caused by the higher frequency cycles, due to the  
331 tower shadow effect. The fatigue model obtains the tidal velocity from equation (1) and the maximum  
332 strain in the blade during that revolution, equation (5). Equation (4), for the observed strain level,  
333 determines the number of cycles to failure. For a particular integration point in the FE model of the  
334 blade, the damage fractions are summed to find the 7-day damage and hence the turbine life, using a  
335 Miner's rule approach [52]:

336

$$337 \quad D_{7-day} = \sum_{k,tide}^{N_{7-day}} \frac{1}{N_{f,k}} + \sum_{j,rev}^{N_{7-day}} \frac{1}{N_{f,j}} \quad (6)$$

338

339 where  $N_k$  is the number of tidal movements during the 7-day period,  $N_j$  is the number of turbine  
340 revolutions during the 7-day period,  $N_{f,k}$  and  $N_{f,j}$  are the numbers of cycles to failure for a given  
341 combination of mean and alternating strain during each tide and revolution, respectively,  $k$  is the  
342 increment of tidal cycles, and  $j$  is the increment of revolutions of the turbine. Hence, the blade life in  
343 years is:

344

$$345 \quad Blade\ life_{years} = \frac{7.38}{365.25 D_{7-day}} \quad (7)$$

346

347 where 7.38 is the exact length of the 7-day period used (1/4 of a synodic month) and 365.25 the length  
348 of a year in days.

349

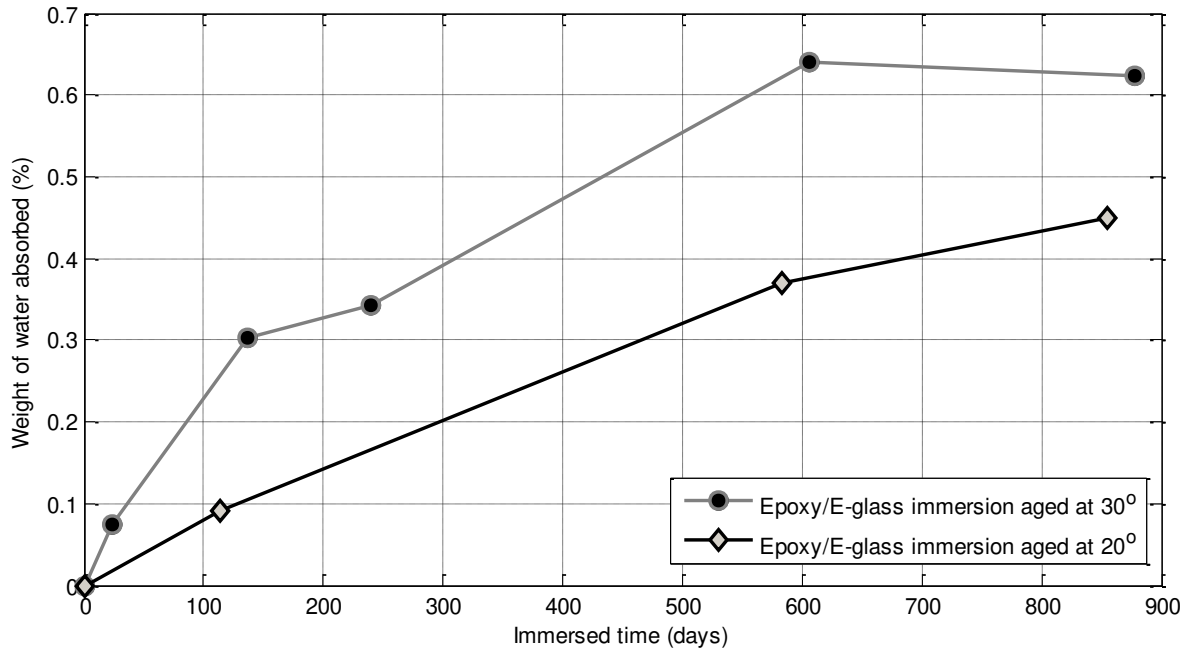
## 350 **5. Results and Discussion**

### 351 **5.1 Water uptake during the immersion-aging period**

352 The percentage weight of water absorbed by the epoxy/E-glass coupons immersed in the ageing  
353 tanks at 30°C and at 20°C (room temperature) is shown in Figure 9. The higher temperature has enabled  
354 the coupons immersed at 30°C to absorb water more quickly and become saturated at approximately  
355 0.60% moisture. In comparison, the coupons at 20°C had not become saturated by the end of the  
356 immersion period (30 months). However, the percentage of moisture absorbed by the epoxy composite  
357 coupons in this study is amongst the lowest reported in the literature [53]. There are two reasons: 1) the  
358 particular polymers are designed for use in immersed applications, , and 2) the laminates from which  
359 the coupons were taken had a low void content (0.22% void content measured in the epoxy/E-glass).

360

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361

362 Figure 9. The weight of water absorbed by Epoxy/E-glass composite specimens during water  
363 immersion ageing.

364

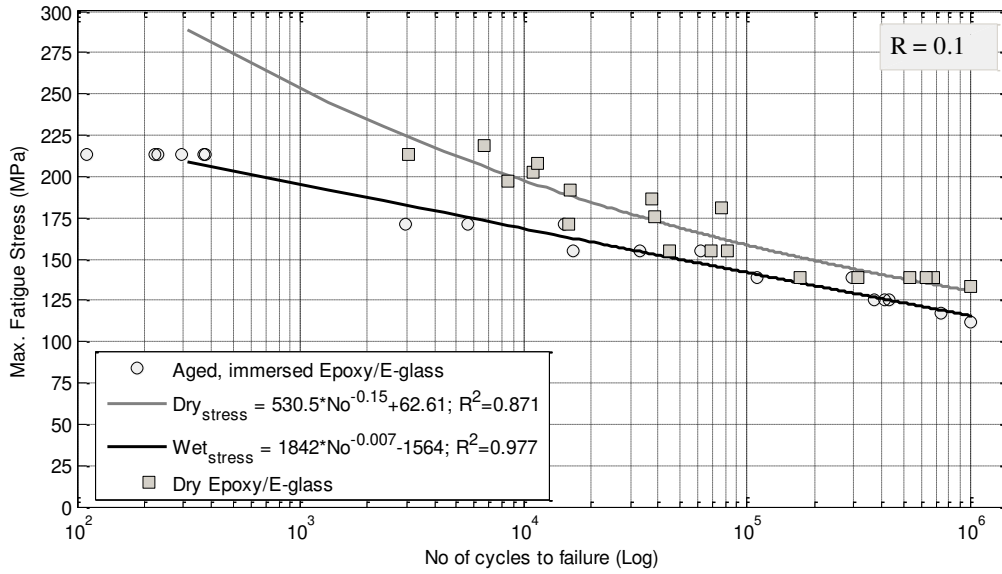
## 365 5.2 Fatigue test results

366 The stress-life fatigue testing results for wet and dry epoxy/E-glass QI coupons in tension-tension  
367 mode ( $R = 0.1$ ) are shown in Figure 10. The dry coupons are tested at room temperature ( $20^{\circ}\text{C}$ ). The  
368 wet coupons immersed for 30 months in  $30^{\circ}\text{C}$  water are tested while immersed in water at  $20^{\circ}\text{C}$ . All  
369 the immersed coupons broke in the zone of immersion within the pouch during fatigue testing. It is  
370 important to keep the coupons wet during testing for the best representation of the immersed failure  
371 condition, as any drying that takes place during the testing tends to be accompanied by a recovery in  
372 strength [54]. The water aged and immersed coupons show a significant decrease in fatigue strength  
373 compared to the dry air aged coupons. The wet aged fatigue strength decreases by 20 to 25% at high  
374 stresses, while the wet aged strength is 8% below the fatigue strength of the dry material at high cycles.  
375 Hence, the effect of water saturation on the fatigue strength of QI GFRP is stress level dependent.  
376 Therefore, the fatigue life reduction of an immersed structure will depend on the spectrum of fatigue  
377 stress cycles it experiences while in service.

378



[Type here]



379

380

Figure 10. Stress-life curves for wet and dry epoxy/E-glass in R = 0.1 fatigue tests.

381

### 382 5.3 Damage model results

383

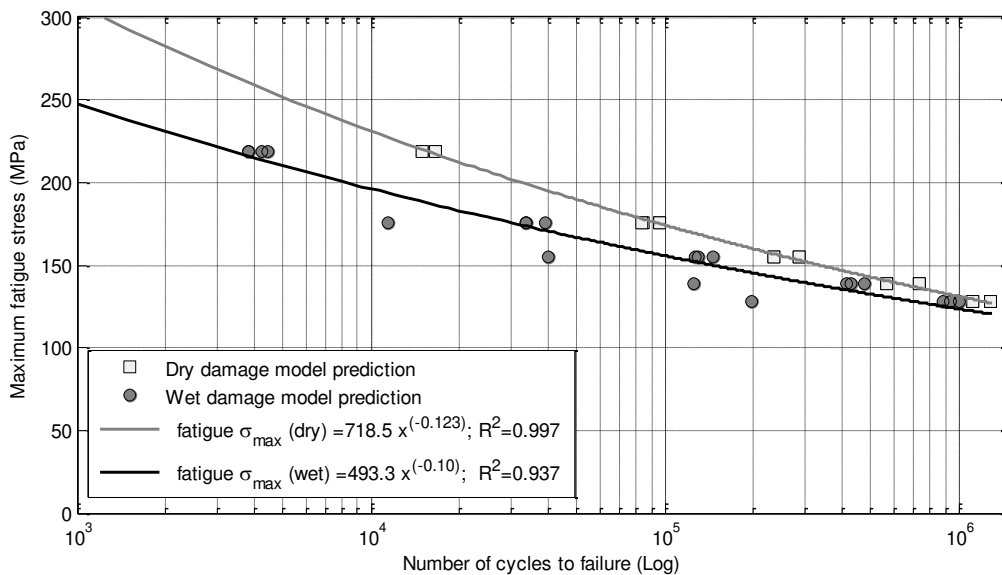
384

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386

387

The SN curves generated using damage model predictions of fatigue life for a range of constant amplitude fatigue stresses in both wet and dry QI epoxy/E-glass laminates are shown in Figure 11. The model successfully predicts the stress dependence of the fatigue strength degradation due to water saturation. There is good agreement between the prediction and the experimental findings: a 20% reduction in fatigue strength due to water immersion at a fatigue life of 1,000 cycles.



388

389

390

Figure 11. Damage model predictions for wet and dry stress-life curves of QI epoxy / E-glass coupons in constant amplitude fatigue.

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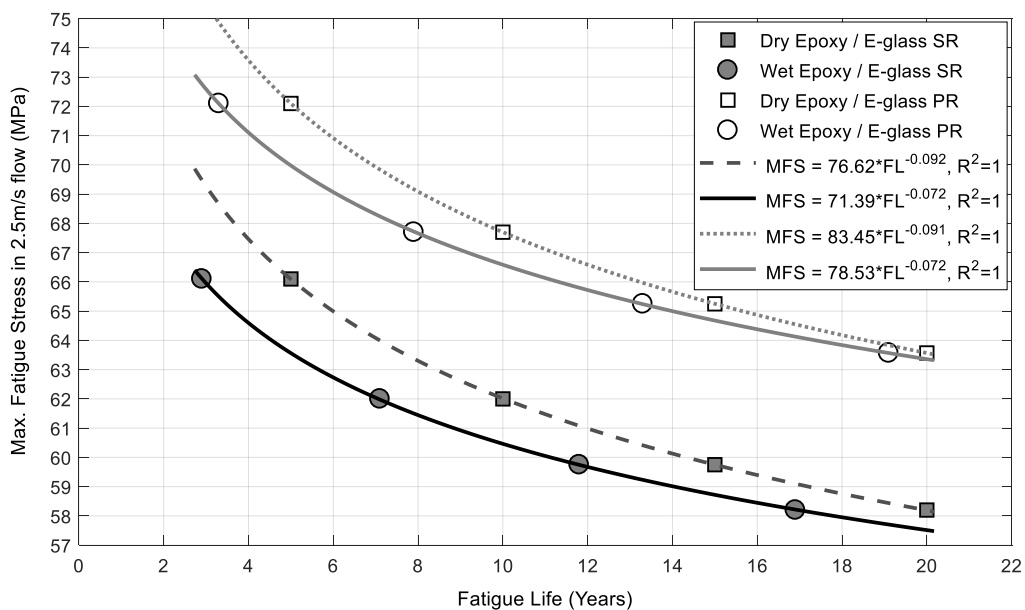
391

#### 392 **5.4 Fatigue life with tidal turbine blade stress spectrum**

393 The predicted fatigue lives of Stall-Regulated (SR) and Pitch-Regulated (PR) TTBs and  
394 corresponding fitting curves are plotted against maximum stress experienced by the blade when the  
395 tidal flow is 2.5 m/s (Figure 12). The results for the PR turbine shows that a blade with a maximum  
396 stress of 72 MPa, which is predicted to have a 5 year “dry” life, will fail, on average, 1.7 years earlier  
397 if the blade laminates are water saturated. If the thickness of the laminate is increased to reduce the  
398 stresses in the blade to 63.5 MPa, the “dry” life becomes 20 years and the “wet” life decreases to 19.1  
399 years when water saturated. This convergence of the wet and dry laminate lives occurs because pitch  
400 regulation limits the maximum stress in the blades. In achieving the longer blade fatigue life (> 15 years),  
401 stresses are limited to levels where the difference in life between wet and dry laminates is very small.

402 The predicted 20 years “dry” fatigue life of an SR TTB is reduced to 17 years if the laminate is  
403 water saturated. Furthermore, if the maximum stress in the blade is 66 MPa, then the predicted “dry”  
404 life is 5 years and the predicted “wet” life is 2.9 years. In general, the life of water saturated blades is  
405 reduced approximately 3 years and 1–2 years for SR and PR turbines, respectively. Hence, the laminates  
406 would need to be 1 to 4% thicker to counteract the effects of water saturation on the blade.

407



408

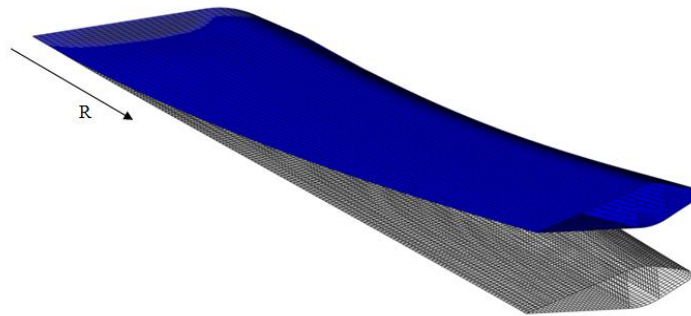
409 Figure 12. Effect of seawater immersion on predicted fatigue life of pitch-regulated (PR) and  
410 stall-regulated (SR) tidal turbine blades.

411

#### 412 **5.5 Blade Finite Element analysis**

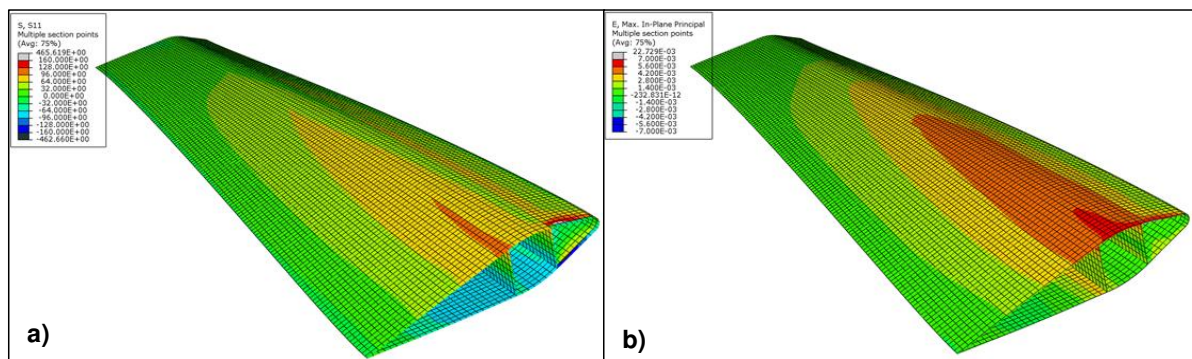
[Type here]

413 Figure 13 shows the deflection obtained using an FEA model of the blade (root area excluded)  
414 exposed to factored loading (safety factor = 2). The deflection of the tip is 334 mm (approx. twice the  
415 blade aerofoil thickness at the tip) and does not introduce any significant error into the structural and  
416 hydrodynamic calculations.  
417



418  
419 Figure 13. The undeformed and deformed shape of the blade at 2.5m/s water velocity, with a  
420 factor of safety of 2.0 applied.

421  
422 As expected, the highest stresses and strains occur near the root, in the spar caps of the structural  
423 beam box section (Figure 14). These are primarily caused by the large (flapwise) bending moments due  
424 to the thrust force.  
425



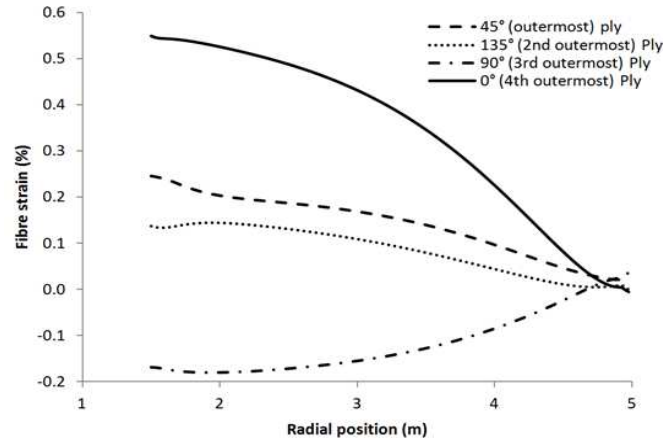
426  
427 Figure 14. a) Fibre direction stresses in 45° surface ply under design load. b) Maximum principal  
428 strains in the outer surface under design load.

429  
430 The maximum bending strains occur on the outer surface and the maximum principal strains in  
431 the middle of the spar cap. Since the surface ply is at 45° to the blade axis, it is necessary to find the  
432 maximum fibre strain in a ply in which the fibres are closely aligned to the load direction in order to  
433 make an appropriate comparison to the experimental fatigue life data. Figure 15 shows a plot of fibre  
434 strains along a section through the midline of the pressure side spar cap extending from near the hub to  
435 the tip of the blade. The fibres in the three outermost plies are at 45°, 135° and 90° to the blade axis

[Type here]

436 which reduces their fibre strains. The 4th ply from the surface has its fibres at  $0^\circ$  to the blade axis (i.e.  
437 aligned with the direction of the principal bending strains). Hence, the maximum fibre strains are  
438 predicted in this ply.

439



440

441 Figure 15. Fibre strains in the four outermost plies at maximum strain locations along blade  
442 operating in 2.5 m/s water velocity.

443

444 The maximum fibre strain in the blade at 2.5 m/s water velocity is used to calculate maximum  
445 fibre strain at any water velocity given the flapwise bending moment at that water velocity and the  
446 assumption of linear elasticity in the blade using Equation 5. By integrating over the tidal pattern  
447 predicted maximum theoretical power levels for both PR and SR turbines, obtained by HDM, the energy  
448 produced by each turbine per year is calculated. In order to compare the PR and SR turbines with respect  
449 to fatigue damage accumulation, the energy output from the PR turbine is matched to that of the SR  
450 turbine. This is achieved by identifying the threshold value of water velocity for pitch control, above  
451 which blade pitch is controlled to give constant flapwise moment (power). Using HDM for the PR  
452 turbine, the threshold water velocity value is found to be 3.05 m/s.

453

## 454 5.6 Blade fatigue life and energy production

455 In order to predict fatigue damage accumulation (Equations 4-6), the moment-velocity  
456 relationships are fitted with polynomial expressions, to allow interpolation with respect to water  
457 velocity. The fatigue model used to analyse the case given in Table 3 predicted a fatigue life of 11.6  
458 years. The fatigue life model can be employed to study the effect of each of the parameters on the blade  
459 fatigue life [23].

460

461

462

Table 3. Parameters for fatigue reference case.

[Type here]

<b>Parameter</b>	<b>Value</b>
Water Velocity	4.0m/s
Neap max. velocity / Spring max. velocity	60%
Factor of safety applied to loads	2
Tower shadow	50%
Control system	PR

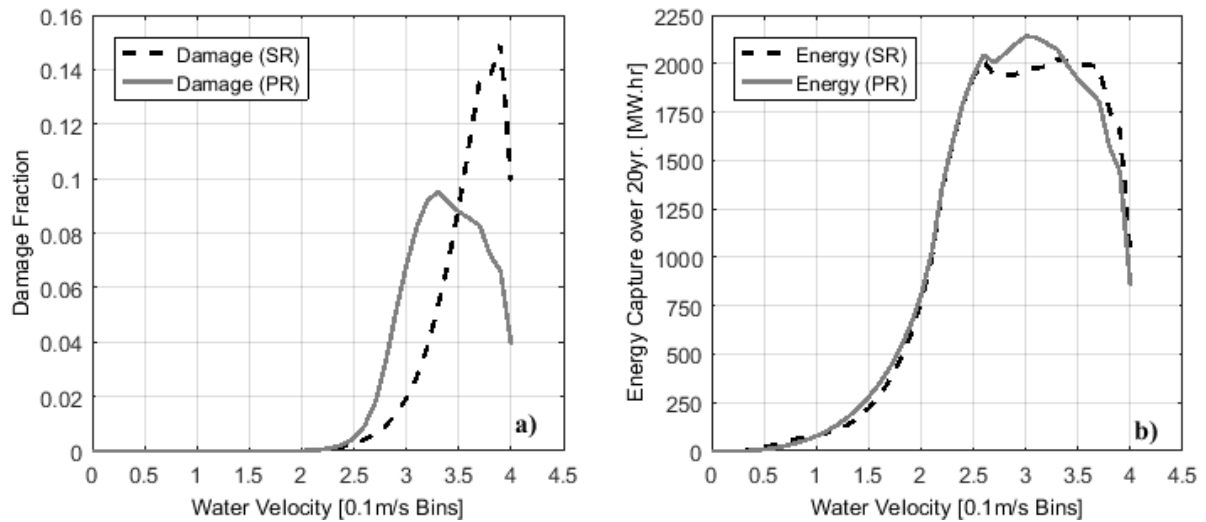
463

464 To compare the fatigue life of SR and PR turbine blades, the fatigue damage fraction for each  
465 rotation of the turbine is sorted into bins for 0.1 m/s increments of water velocity, as shown in Figure  
466 16a (note that the area under each curve, which represents total accumulated damage over the blade  
467 lifetime, is equal to 1.0). The SR blade accumulates a lot of damage at high water velocities, due to the  
468 higher bending strains, whereas for the PR blade, the damage at high velocities is significantly lower,  
469 due to the lower bending strains. Most of the PR blade damage is sustained at medium water velocities,  
470 due to the greater number of operational hours in this regime. The fatigue model predicts that a blade  
471 which survives 20 years on the PR device will only survive 12.8 years on the SR device. On the other  
472 hand, if the laminates in the spar cap of the blade are increased by just 10% it would survive 22 years  
473 on the SR machine. An increase in water velocity will not reduce blade life because the blades will  
474 feather at maximum power either way.

475 Figure 16b shows energy capture for different control systems (note: the stall characteristics and  
476 the pitch regulating point of the corresponding blades were chosen to give the same energy capture over  
477 the 20-year design life). If the maximum or average speeds at the turbine site do not match the design  
478 speed of the turbine it can have significant impacts on energy production or the life of SR turbine. A  
479 10% decrease in the velocity of tidal flows will reduce energy production per year by 20%, whereas a  
480 10% increase in tidal speed will reduce life from 20 to 12.1 years. In this situation, the PR turbine would  
481 not experience the same reduction in energy production as the SR turbine. There will be some loss in  
482 energy produced because the average will be lower, but looking at the bins data it can be seen that only  
483 1/4 of the energy is produced before the pitch regulation point and will be affected by the loss in speed,  
484 above this point the turbine will make the same power either way.

485

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486

487 Figure 16. Stall- and pitch-regulated tidal turbine blades: a) damage fraction and b) energy  
488 capture for different water velocities.

489

## 490 6. Conclusion

491 A preliminary design and fatigue life assessment methodology for composite tidal turbine blades  
492 (TTBs) is used to compare the predicted fatigue life of pitch-regulated (PR) and stall-regulated (SR)  
493 tidal turbine devices. The immersion of GFRP materials in 12°C seawater for a period of 20 years is  
494 simulated by immersing specimens in warm water (30°C) for approximately 2.5 years, showing that the  
495 coupons absorbed relatively small amounts of water (0.2% to 0.6%) due to the characteristics of chosen  
496 material and fabrication procedures adopted. The fatigue strength of the aged-water-immersed material  
497 is stress-level dependant and is significantly reduced, showing 20% and 8% reduction for a fatigue life  
498 of  $10^3$  cycles and  $10^6$  cycles, respectively, in comparison with the dry samples.

499 It was found that the life of water-saturated blades is reduced by about 3 years and 1–2 years for  
500 SR and PR turbines, respectively. In order to neutralise the effect of the water saturation, the laminates  
501 need to be thicker by 1-4%. The study of SR and PR TTBs based on same energy capture over the 20-  
502 year design life found that blades on an SR tidal turbine need approximately 10% thicker laminates than  
503 those on a PR turbine for a given design fatigue life.

504 The results of this study were encouraging for the future development of more advanced fatigue  
505 analysis and design methodologies for tidal turbine blades, which will include more structural details  
506 (e.g. root connections, stress concentration at ply drops and holes, etc.), more comprehensive material  
507 modelling (e.g. modelling of individual plies with associated orientations) and more realistic water  
508 velocity conditions (e.g. effect of turbulence and wave-loading).

509

510 **Acknowledgements**

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515

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638 2014, pp. 165-177.
- 639
- 640

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## 641 **Appendix 1: Blade element momentum theory (BEMT)**

642

643 For steady state operation the fluid forces on the blade inside any stream tube must equal the  
644 momentum lost from that stream tube:

645

$$646 F_A = \left(\frac{2\dot{M}}{N}\right) (V_t - U_d(R)) \quad (1A.1)$$

647

648 where  $F_A$  is the sum of the axial components of the lift and drag forces on the blade element  
649 inside the tube,  $N$  is the number of blades,  $\dot{M}$  is the mass flow rate through the tube,  $U_d$  is the axial  
650 water velocity at the rotor disk, and  $R$  is the radius of the disk. Within each stream tube the tangential  
651 components of the lift and drag forces sum to give a torque producing force and this imparts angular  
652 momentum to the blade. The angular momentum is balanced by an angular momentum of the fluid in  
653 the opposite direction, causing wake rotation. With the assumption that the relative resultant water  
654 velocities are constant, axial and rotational velocity at the disk can be calculated:

655

$$656 U_d = \frac{V_t}{\pi} \sqrt{1 + \gamma^2 + \zeta - \gamma \sqrt{(1 + \gamma^2 - \zeta^2)}} \quad (2A.1)$$

657

$$658 V_d = U_d \frac{\sqrt{1 + \gamma^2 - \zeta^2} - \gamma}{1 - \zeta} \quad (3A.1)$$

659

660 where  $V_d$  is the tangential velocity of the fluid in a stream tube at the rotor disk,  $\zeta$  is the fraction  
661 of axial velocity remaining at the downstream exit of the stream tube, and  $\gamma$  is the local speed ratio  
662 given by:

663

$$664 \gamma = \frac{\omega_b R}{V_t} \quad (4A.1)$$

665

666 where  $\omega_b$  is the angular velocity of the blade,  $R$  is the radius, and  $V_t$  is the free stream velocity  
667 of the fluid. The blade also has a tangential velocity  $V_b$  at the stream tube due to its angular velocity:

668

$$669 V_b = \omega_b R \quad (5A.1)$$

670

671 The relative velocity at the rotor:

672

[Type here]

673 
$$V_r = \sqrt{U_d^2 + (V_d + V_b)^2} \quad (6A.1)$$

674

675 and the angle between the relative velocity and the plane of the rotor is:

676

677 
$$\theta = \tan^{-1} \frac{U_d}{V_d + V_b} \quad (7A.1)$$

678

679 The lift force,  $L$  (perpendicular) and drag force,  $D$  (parallel to the fluid velocity), generated by a  
680 stream tube on the blade section inside it:

681

682 
$$L = \frac{1}{2} C_L V_r^2 \rho S \Delta R \quad (8A.1)$$

683

684 
$$D = \frac{1}{2} C_D V_r^2 \rho S \Delta R \quad (9A.1)$$

685

686 where  $C_L$  and  $C_D$  are the coefficients of lift and drag for the aerofoil selected [33],  $\rho$  is the density  
687 of the fluid,  $S$  is the chord length of the blade at the stream tube radius and  $\Delta R$  is the radial thickness of  
688 the stream tube. The axial and tangential force is:

689

690 
$$F_A = L \cos \theta + D \sin \theta \quad (10A.1)$$

691

692 
$$F_C = L \sin \theta + D \cos \theta \quad (11A.1)$$

693

694 where  $F_A$  is the axial (thrust) force and  $F_C$  is the tangential (torque) force acting on the blade by  
695 observed stream tube. According to the axial induction factor, maximum power output corresponds to:

696

697 
$$U_d = \frac{2}{3} V_t \quad (12A.1)$$

698